Noise and indoor air quality - HVAC noise characteristics

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Abstract

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Noise from Heating Ventilation and Air Conditioning (HVAC) systems in residential buildings have an influence on the residents´ well-being. The characteristics of HVAC noise have been investigated as to recommend typical spectra for the design of a listening test using sound recordings. An HVAC system layout is developed to represent a system in a residential building where variations of the noise spectrum are possible without the use of unrealistic conditions. From a spectrum shape perspective the main parameter of the system is the balance between noise originated from the heat recovery unit and the duct component related noise. The receiving room spectrum is here mainly modified by the use of different silencers close to the heat recovery unit and the balance between the pressure loss of the air terminal device and a nearby damper. Modifications of the heat recovery unit total airflow and the inflow properties to the air terminal device are also investigated. Variations at both low and high frequencies are possible, at a constant total A-weighted sound pressure level, using these system properties.

Key words: HVAC, ventilation, residential buildings, flow generated noise

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Sammanfattning

Buller från ventilationssystem (HVAC) i bostadshus har en inverkan på de boendes välbefinnande. Ventilationssystemets olika bulleralstrande komponenter har här undersökts för att rekommendera typiska bullerspektra för användning vid ett lyssningstest. Ett ventilationssystem har dimensionerats för att representera ett typiskt system i ett bostadshus där variationer av bullerspektra är möjligt utan att använda orealistiska inställningar. Ljudinspelningar kommer vid ett senare skede genomföras för detta system. Ventilationssystemets luftflöden har definierats utifrån en analys av föreskrivna krav på luftomsättning och genomsnittliga bostadsytor. Riktvärdet för mottagarrummets ljudtrycksnivå är bestämt utifrån befintlig kravställning för ett typiskt vardagsrum. Även efterklangstiden i mottagarrummet är vald utifrån ljudabsorptionen i ett vardagsrum.

Huvudparametern för bullervariationerna är balansen mellan ljud orsakat av ventilationsaggregat i jämförelse med ljud alstrat av kanalkomponenter. Mottagarrummets bullerspektrum har huvudsakligen varierats genom användning av olika aggregatljuddämpare samt modifiering av balansen mellan tilluftsdonets tryckfall och tryckfallet över ett närliggande spjäll. Variationen av tilluftsdonets tryckfall kan ses som ett resultat av lämpligheten i ventilationssystemets injustering och valet av ljuddämpare kan ses som en följd av den akustiska planeringen. Variationer av aggregatets totala luftflöde och egenskaperna hos tilluftsdonets inflöde undersöks också. Det totala luftflödet är en följd av storleken och antalet hushåll som ventilationsaggregatet försörjer. Inströmningsegenskaperna påverkas av tillgängligt utrymme mellan de olika kanalkomponenterna såsom om det finns en böj innan tilluftsdon. Variationer av både låga och höga frekvenser vid en konstant A-vägd ljudtrycksnivå är möjliga med hjälp av de beskrivna systemegenskaperna.

1 Introduction

The project “Noise and indoor air quality - considerations for residents' well-being” aims to investigate the experience of noise from Heating Ventilation and Air Conditioning (HVAC) systems in residential buildings. The experienced disturbance will be evaluated by listening tests using sound recordings of HVAC noise. Today, the noise legislation for residential buildings and the HVAC industry are focused on weighted total noise levels. The listening tests will evaluate the benefit of different psychoacoustic measures, e.g., sharpness and roughness, for the use in acoustic planning of HVAC systems. To make a suitable choice of reference noise levels and spectra the noise generating components of residential HVAC systems were investigated as here presented. The acoustic analysis is limited to spectrum shape and level. The aim is to answer two questions:

1. What variations in HVAC noise, and especially in noise spectrum, do we have in residential buildings?
2. How can we design a measurement setup to demonstrate these variations under realistic conditions?

To answer the first question the main components of the HVAC systems are analysed as described in chapter 2. The second question is investigated in chapter 3 where the design of the chosen measurement setup and the assumptions around it are described. Chapter 4 includes the measurement results from the HVAC system setup and recommends noise spectra for the sound recordings.

2 HVAC noise in residential buildings

There are several sources and transmission paths for HVAC noise in a residential building. Dependent on the layout of the duct system and the choice of components the dominant noise source will vary. Recommendations for the noise design of a HVAC system can be found in e.g. VDI 2081 [VDI 2081] or ASHRAE Handbook for HVAC Applications [ASHRAE]. A corresponding review in Swedish is “Ljuddimensionering av ventilationssystem” [Nyman och Danielsson]. A common HVAC system layout and noise characteristics of the system components are here described.

2.1 The HVAC system

For the scope of this project a HVAC system where a Heat Recovery Unit (HRU) provides supply and extract airflows together with heat recovery and air filtration is assumed. In residential buildings a common choice is to have one HRU for each household, the benefit being the opportunity to control the HVAC parameters independently of other households in the building. The option of one larger HRU serving multiple households is also considered.

The HVAC duct system is divided into a supply air part and an extract air part as presented in figure 1. The supply air normally provides an inflow of fresh air to bedrooms and living rooms. The air is extracted from bathrooms, wardrobes and the kitchen by the extract air system. In each room one or more Air Terminal Devices (ATD) are distributing/extracting the air to/from the room. Depending on the air distribution strategy in the room different designs of the ATDs are available. To control the amount of air provided/extracted to each room the pressure drop of the ATD can be set. An alternative is to install a damper in each branch of the duct system and thereby balance the airflow to/from the rooms.

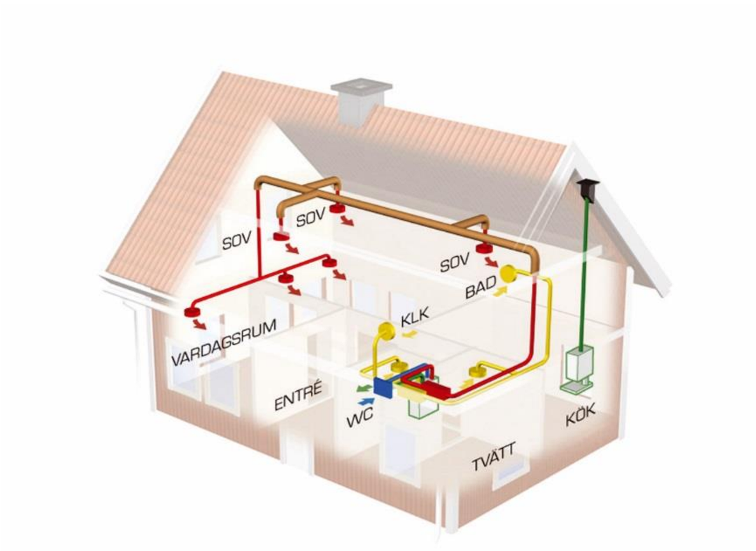

Figure 1. Common HVAC system layout in residential buildings. Picture used with permission from Fläkt Woods

To improve the energy efficiency of the HVAC system the heat recovery component in the HRU is transferring the heat in the extract air to the supply air. If the fresh air still needs to be heated to reach a suitable temperature an additional heater is installed. The fan wheel, generating the airflow, is together with the fan motor another important component for the energy efficiency. The specific fan power is specifying the electrical power needed by the fan wheel and motor to provide a certain amount of air.

2.2 Fan generated noise

When generating the airflow the fan is causing flow generated noise into the duct. In addition to this airborne noise the fan is generating structure borne noise due to the vibrations transmitted to the structure. The structure borne noise can be limited by a thorough balanced fan wheel and vibration isolators. No analysis of the structure borne noise in the building is here presented.

For residential buildings a fan wheel generating the air flow in a radial direction is today the common choice [Nyman and Danielsson]. The blades of the fan wheel can either have a forward or a rearward angle in reference to the rotation direction. Different aspects of energy efficiency and noise generation must be taken into account when selecting the type and size of fan wheel.

The flow generated noise can be divided into a broadband noise component and a tonal component. The fan generated turbulence and the shedding of the turbulence are the main sources of the broadband component [VDI 2081]. The level and the spectrum of the broadband noise will differ depending on the fan model and the duty point of the fan. In a similar way the frequency and the amplitude of the tonal components will vary. The main cause of the tonal component is the periodic passing of the fan wheel blades to space stable parts in the fan wheel mounting [VDI 2081]. The frequency of this so called blade passing frequency and its overtones can be determined from the multiples of Eq. (1)

$$f = \frac{n \times z}{60} \qquad (1)$$

where n is the rotation speed and z the number of fan wheel blades [VDI 2081]. There are different definitions of tonality to determine the extent of tonal components in the broadband noise e.g. ISO 1996-2:2007 annex C and D [ISO1996-2]. The simplified method in annex D consists of a comparison of third octave band levels between one band to the adjacent two bands. If the band exceeds both adjacent band by some constant level the sound is considered as tonal.

The noise level of the fan is normally given in the product data of an HRU as a function of airflow and available static pressure. The spectrum is often given in octave band levels for the band centre frequency range 63 – 8000 Hz.

2.3 Noise from duct system components

The duct system components will attenuate the noise from the fan and for each component add flow generated noise. The final noise level out from the air terminal devices is normally approximated by stepwise calculation starting with the sound power level from the HRU. The component attenuation is subtracted from the incoming sound power level and the corresponding flow generated noise is added resulting in the incoming sound power level to the next component. An example for the project HVAC system is presented in Appendix A. The main limitation of the described calculation approach is the assumption of reflection free conditions for each component neglecting, especially at low frequencies, the interaction between sound source and duct system [Rämmal and Åbom]. For a more detailed HVAC system calculation a sound wave based model rather than a sound power based model is recommended.

Specific product data for correct duty points can be used in the duct system calculations if available. Alternatively, if the specific products are not determined, noise predictions e.g. semi-empirical scaling laws [Kårekull] can be used. A noise prediction example comparing the use of semi-empirical scaling laws to product data for the project HVAC system design is presented in Appendix B.

2.3.1 Straight ducts and bends

The focus on energy efficiency in HVAC systems has resulted in generally lower flow velocities in ducts. The normal consequence is negligible contributions of the flow generated noise from straight ducts. Eq. (2) determines the A-weighted sound power level of flow noise from straight ducts [VDI 2081]

$$L_{WA} = -25 + 70\log(v) + 10\log(S) \qquad (2)$$

where v is the air velocity in duct and S is the duct cross section area. For example the sound power level is lower than 15 dB(A) for ducts with a diameter of less than 200 mm when keeping the flow velocity below 6 m/s. In a similar way bends will generate low noise levels at moderate flow speeds. A 90° bend in a circular duct will generate less than 37 dB(A) for duct diameters equal or less than 160 mm and a flow velocity below 6 m/s. Table 1 presents the sound power spectrum at 6 m/s for a 90° bend [Nyman and Danielsson]. The corresponding sound reduction of straight ducts and bends are presented in table 2 and 3

Table 1. Generated octave band sound power levels of a 90° bend in a circular duct at an air velocity of 6 m/s [Nyman and Danielsson]

f [Hz]	63	125	250	500	1000	2000	4000	8000
Lw [dB]	54	45	40	35	24	16	9	2

Table 2. Sound attenuation [dB/m] for straight circular duct of diameter 0.1-0.2m [VDI 2081]

f [Hz]	63	125	250	500	1000	2000	4000	8000
D=160mm	0,1	0,1	0,15	0,15	0,3	0,3	0,3	0,3

Table 3. Sound attenuation of a circular 90° bend in duct of diameter D [VDI 2081]

f [Hz]	63	125	250	500	1000	2000	4000	8000
D=100mm	0	0	0	0	1	2	3	3
D=160mm	0	0	0	1	2	3	3	3

2.3.2 Dampers and junctions

Junctions in combination with dampers are used in the duct system to divide the airflow between the different parts of the building. The flow noise generation of a junction will depend on the proportion of the airflow split, the duct diameters and the inner radius of the junction rounding's. A noise prediction calculation for the junction flow noise is presented in VDI 2081. The sound level reduction of a junction is normally determined by the proportion between the duct area of interest to the total area of the junction ducts as presented in figure 2.

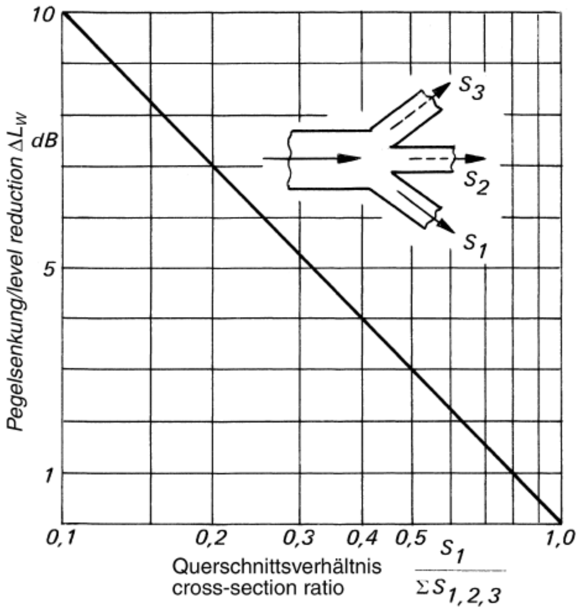

Figure 2. Level reduction at junctions. Picture from VDI 2081

There are different designs of dampers e.g., butterfly and iris layout as presented in figure 3. The flow noise generation of dampers can be found in product data as a function of airflow and pressure drop or calculated from noise prediction models. The sound attenuation of a damper is generally neglected in the noise calculation for residential buildings.

Figure 3. Butterfly(left) and iris (right) dampers. Pictures used with permission from Fläkt Woods.

2.3.3 Silencers

The purpose of a silencer is to attenuate the duct noise level introducing a pressure drop as low as possible. Normally the silencer is combining an effect of reflection using a duct diameter change and absorption using the silencer filling material e.g. mineral wool. Equivalent to straight ducts the flow noise generation is low for a silencer at moderate air velocities where the corresponding duct cross section area is clear. Silencers introducing curtains, filled with insulation material, in the duct cross section area do normally have a higher sound attenuation in combination with an increased pressure drop and a correspondingly increased flow generated noise level. The silencer length is normally proportional to the sound attenuation, especially in the low frequency range.

2.3.4 Air terminal devices

The air terminal device is, by distributing the duct airflow into the receiving room or extracting the room air, introducing a component pressure drop, a sound attenuation and a flow generated noise level. There is a wide range of ATD designs and strategies for an optimal air distribution/extraction performance. The noise characteristics can be found in product data or approximated by HVAC guidelines. The sound attenuation is dominated by the end reflection effect determined from e.g. SS-EN ISO 5135:1997

2.4 Sound radiation in rooms

The dominating HVAC noise sources are radiating into the room from the positions of the ATDs, excluding the structure borne noise. To determine the sound pressure level in different positions of the room, except from positions in the direct field i.e. close to the noise sources, the reverberant sound field is normally assumed diffuse. To be considered as diffuse the sound field needs to meet two conditions [Hodgson]:

1. For all positions in the reverberant sound field waves are to be incident from all directions with equal intensity and having a random phase relation.
2. Every position is to have the same reverberant sound field.

A room approximately meets these conditions if [Kleiner and Tichy]:

- The room average sound-absorption coefficient is less than 0.3
- The sound-absorbing surfaces are evenly distributed over the room
- $l_{max} < 1.9V^{1/3}$ where V is the room volume and l_{max} is the longest straight line that fits within the boundary of the room.

For simplicity, even if these conditions are normally not met [Hodgson, VDI 2081] the sound field is often considered diffuse. In the proximity of each ATD the direct field will dominate the sound pressure level. In the reverberant field, the wall reflections will dominate and the room absorption properties will be of importance. Using the diffuse field assumption, the sound absorption of the room is defined as the difference between the mean sound pressure level, L_P, and the sound power level of the noise sources, L_W. The sound absorption can be determined from the equivalent sound absorption area, A, according to Eq (3) or the reverberation time, T, and the room volume, V, according to Eq (3) and (4) [VDI 2081].

$$L_P - L_W = 10log\left(\frac{4}{A}\right) \qquad (3)$$

$$A = 0.163\frac{V}{T} \qquad (4)$$

Since the assumption of a diffuse sound field often can be questioned there is a need of more detailed calculation methods. Different versions of ray tracing models and empirical models have been developed. One standardized method is described in VDI 3760 having the ambition of a low computational cost. The disadvantage is that only the mean absorption of walls is taken into account and no source directivity or diffraction effect over large structures are included.

2.5 Legislation and average size of Swedish residential buildings

The air exchange in a Swedish residential building is legislated by The Public Health Agency (Folkhälsomyndigheten). Among other rules, e.g. air exchange per person, the air exchange for every building square meter shall be at least 0.35 l/s [FoHMFS 2014:18]. The corresponding sound pressure level for noise from installations, e.g. HVAC systems, in residential buildings is in Sweden regulated by Boverket´s building regulations [BBR]. The noise level is regulated using four different measures:

- Equivalent A-weighted sound pressure level
- Maximum A-weighted sound pressure level
- Equivalent C-weighted sound pressure level
- Equivalent level when impulses, tonality or other noise variations are clearly audible.

The type of usage determines the noise criteria of the room which results in what measures that are specified. Three different examples are presented in table 4.

Table 4. Noise level criteria in residential buildings [BBR]

	L_{pAeq}	L_{pCeq}	L_{pAFmax}	L_{pAeq} (e.g. for tonality)
Livingroom	30	-	35	25
Bedroom	30	50	35	25
Kitchen	35	-	40	30

As an alternative measure for the equivalent C-weighted level low frequency third octave band limits can replace the regulated C-weighted level [BBR]. Limit levels are regulated by the public health agency according to table 5

Table 5. Equivalent third octave band sound pressure levels [FoHMFS 2014:13]

f [Hz]	31.5	40	50	63	80	100	125	160	200
L_{eq} [dB]	56	49	43	42	40	38	36	34	32

The average number of persons per household was in 2015 for Sweden 2.7 in a one- or two dwelling buildings [SCB]. In multi-dwelling buildings the corresponding figure is approximately 2. The average number of square meters per person was in 2015 for Sweden 46 m^2 for one- or two dwelling buildings and approximately 35 m^2 for multi-dwelling buildings [SCB]. The resulting average size is 124 m^2 for one- or two dwelling buildings and 70 m^2 for multi-dwelling buildings.

3 Measurement setup

For the HVAC noise recordings a realistic system design, components and duty points need to be selected. The measurement setup should be well defined and easy to verify but also easy to modify for the different recording settings. The assumptions around the chosen measurement setup are described in chapter 4.1. The chosen layout is presented in section 4.2 and the selected components in section 4.3. The different measurement methods used are described in section 4.4.

3.1 Scope of HVAC system setup

Residential buildings are built in many sizes offering different amount of living space. As legislated, the required air exchange rate can be defined by the number of square meter living space. Regarding one-dwelling buildings the HVAC system will serve one household and for multi-dwelling buildings both one and many households can be served. For the sound recordings a total airflow of 70 l/s is chosen as the standard case. The airflow corresponds in a one dwelling building to 200 m^2 where the minimum air exchange of 0.35 l/s per m^2 is chosen or 140 m^2 if the air exchange is 0.5 l/s per m^2. For apartments, the setup can be one apartment of equivalent area as the one dwelling building or two and three apartments with a corresponding area of half or one third. The examples of possible alternatives are presented in table 6. The range of square meters in table 6 includes both the average one dwelling building of 124 m^2 and the average household size of 70 m^2 in a multi-dwelling building as described in section 2.5.

Table 6. Corresponding area per household for an airflow of 70 l/s

Type of residential building	Area [m^2] per household 0.35 l/s per m^2	Area [m^2] per household 0.5 l/s per m^2
One dwelling building	200	140
1 apartment	200	140
2 apartments	100	70
3 apartments	67	47

For the specific room where the sound recording shall be recorded an airflow of 20 l/s is chosen. The airflow corresponds to an area of 60 m^2 where the air exchange is 0.35 l/s per m^2 and 40 m^2 for 0.5 l/s per m^2. For a larger house the same area could correspond to a large living room to which the air is both supplied and extracted. For a smaller house or apartment the area could correspond to the combination where air is supplied to a living room and extracted from the kitchen or a bathroom. The sum of the areas of both rooms is then to be compared to 60 m^2 and 40 m^2 e.g. living room of 40 m^2 or 25 m^2 and kitchen of 20 m^2 or 15 m^2. Table 7 presents the examples.

Table 7. Corresponding area for an airflow of 20 l/s

Type of room(s)	Area [m^2] per room 0.35 l/s per m^2	Area [m^2] per room 0.5 l/s per m^2
Living room	60	40
Living room + kitchen/bathroom	40 + 20	25+15

To simplify the measurement setup only supply air is considered corresponding to the ventilation case where the air is extracted in another room than supplied to. An assumption of only one supply air terminal device in the receiving room is also made.

A living room reverberation time is around 0.5 s [Buller] i.e. 0.4-0.6 s. A room sound absorption of approximately 8dB, i.e. equivalent to 25 m^2 sound absorption area, is chosen. The corresponding room size for a reverberation time of 0.6 s is a volume of approximately 100 m^3 and for 0.4 s a room volume of 62 m^3. Two common room heights in Sweden are 2.4 and 2.7 m. Table 8 presents a calculation of corresponding room floor area. The calculated areas correspond well to the room sizes defined for the characteristic households.

Table 8. Room floor area from reverberation time assumption

	Rev. time [s]	Abs. [dB] (m^2)	Volume [m^3]	Area [m^2]
Room height 2.7m	0.4	8 (25)	62	25
Room height 2.7m	0.6	8 (25)	100	40
Room height 2.4m	0.4	8 (25)	62	40
Room height 2.4m	0.6	8 (25)	100	60

Additionally the following assumptions are chosen for the acoustic properties of the HVAC system setup.

- The noise legislations of a living room are considered i.e. a room for social gatherings but not for sleeping.
- Structure borne noise is excluded from the analysis
- Whistling in system components, i.e. flow induced instabilities, is excluded when possible.
- No other installation noise sources or e.g. traffic noise is included.

3.2 Layout of HVAC system setup

From the assumptions described in section 3.1 an HVAC system setup is designed. The main parts are a heat recovery unit, a duct system and a receiving room as presented in figure 4. Two dampers are installed to control the system pressure drop and the distribution of air between the main duct and the duct to room. An extra additional pressure drop, consisting of a butterfly damper, is also introduced at the end of the main duct to enable a larger pressure drop. The main duct static pressure and the ATD static pressure drop are measured. The pressure loss coefficient of the ATD can be adjusted.

Three silencers are possible to introduce denoted as A, B and C. Silencer A will attenuate the flow generated noise from the room duct damper and is therefore chosen to be always present in the measurement setup. Another argument for the presence of silencer A is to avoid cross talk between rooms. Silencer B will adjust the amount of HRU noise to the receiving room by modified length or by being completely removed. Silencer C will decrease the noise level introduced to the room of the duct system to avoid noise transmission to the receiving room. The HVAC system is designed as to have a noise level in the receiving room close to but lower than 30 dB(A).

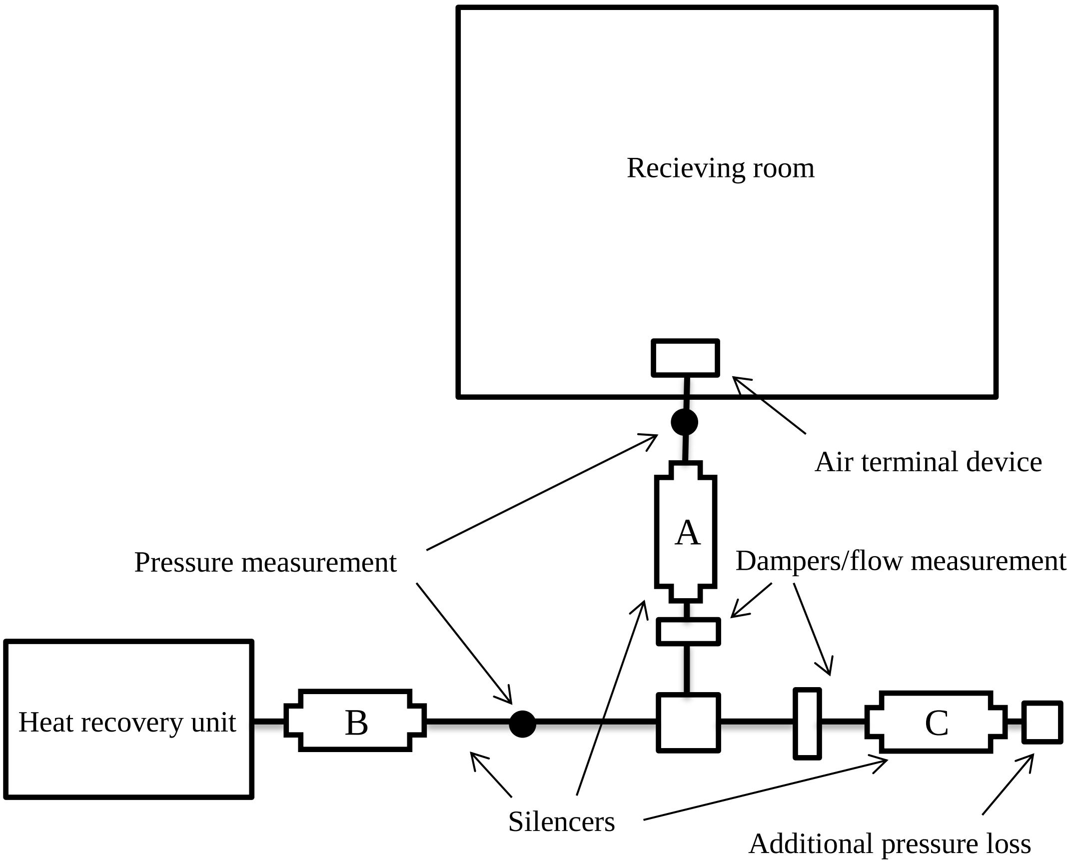

Figure 4. Layout of the HVAC system setup

The following variations are possible for the HVAC system measurement setup:

- Different configurations of silencers
- Different configurations of the pressure drop distribution between duct system components e.g. between ATD and damper
- Increased air flow to receiving room
- Increased air flow of HRU
- Introduction of bend close to ATD

3.3 Chosen components

For the final recordings three HRU, from the three manufacturers involved in the project, are chosen. Information regarding design and performance can be found on the suppliers' webpages [Fläkt Woods, Systemair, REC-Indovent]. Table 9 presents the different HRU:s and figure 5 presents pictures of the products. For the measurement results in this report only measurements for Fläkt Woods RDAS are conducted.

Table 9. Chosen heat recovery units

Supplier	Product name	Maximum airflow [l/s]
Fläkt Woods	RDAS	>100
REC - Indovent	RT Blue4	>100
Systemair	VTR 500	>100

Figure 5. Heat recovery units from left: VTR 500, RT Blue4 and RDAS

The chosen ATD is presented in figure 6 and is manufactured by Fläkt Woods under the name STQA. A benefit of the design is an approximately equivalent noise generation mechanism independent of the set ATD pressure drop. The pressure loss of the ATD can be adjusted by covering a number of rows of hole. The setting is defined from the number of open rows of holes. For the tests 3 different settings are used corresponding to 7 (h7), 10 (h10) and 14 (h14) number of open hole rows.

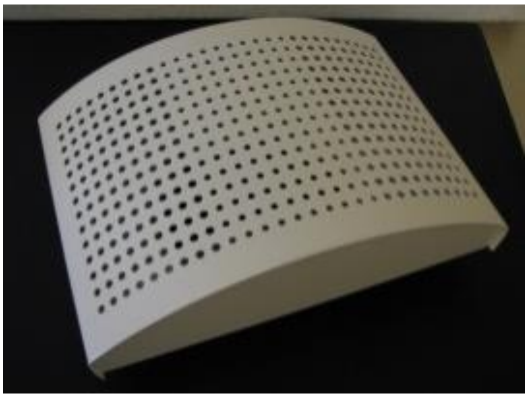

Figure 6. Air terminal device STQA

The design of the dampers is presented in figure 3. Two iris dampers are used; one in the main duct and one in the duct to the receiving room. A butterfly damper is used for the extra pressure drop in the end of the main duct. The chosen silencers are circular with 50mm glass wool. Lengths of 300, 600 and 900 mm or combinations of these are used. The main duct consists of a circular duct of 160 mm diameter. The corresponding air velocity in duct is 3.5 m/s for an airflow of 70 l/s. A circular duct of 100 mm diameter is used for the duct to the receiving room. For an airflow of 20 l/s the air velocity is 2.5 m/s. The distance between the duct components were always more than 10 duct diameters.

3.4 Measurement methods

The measurements have been conducted at the Fläkt Woods laboratory in Jönköping, Sweden. Three physical measures have been conducted: sound pressure level, airflow and pressure drop.

To verify the legislated noise levels BBR refers to SS-EN ISO 10052:2004 and SS-EN ISO 16032:2004. Since the recordings will be taken at one measurement position in the occupational zone of the room the measurement method applied here refers to SS-EN ISO

3741:1999 normally used for measurements in reverberation chambers. Since extra absorption is introduced into the reverberation chamber, the measurement does not fulfil the requirement of an adequate reverberant sound field. The measurements can be seen as an average of the sound pressure level in the occupational zone of the room.

For the determination of the ATD sound power and the performance of dampers a measurement fulfilling the requirements of SS-EN ISO 5135:1997, i.e. SS-EN ISO 3741:1999, was used. The measurement setup is presented in figure 7. The airflow was determined according to SS-EN ISO 5167:2003 and the static pressure drop according to the recommendations in the same standard. Airflow for the HVAC system measurement setup was determined by component measurements of the ATD and damper according to the setup in figure 7. The airflow was then verified in the HVAC system measurement setup by pressure drop measurement of the ATD and the damper. The static pressure in duct was determined by four circumferential duct wall positions.

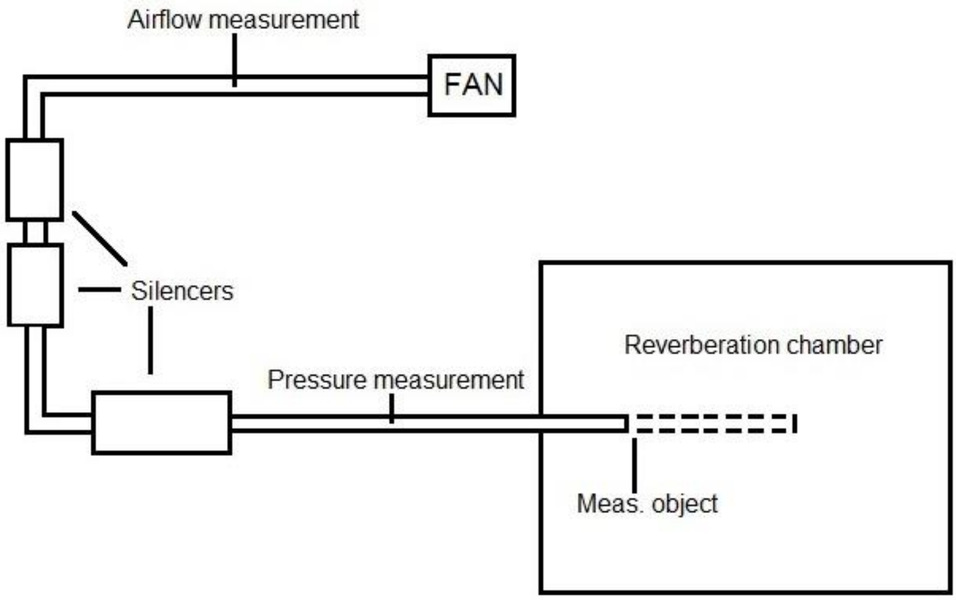

Figure 7. Measurement setup for airflow, static pressure drop and sound power measurements

4 Measurement results

The measurement results aim to verify the performance of the HVAC system components and to evaluate the resulting spectra in the receiving room for the different system variation possibilities. Verifications are presented in section 4.1 for the ATD and dampers. The evaluation of the receiving room spectra is found in section 4.2.

4.1 Air terminal device and damper performance

The performance of the chosen ATD was verified using the measurement method described in section 3.4. For consistency 3 ATD:s of the same model were used for all measurements where each copy always had the same setting. The measured airflow and static pressure drop for the different ATD settings are presented in table 10. The corresponding sound power spectra are presented in figure 8. Effects from the background noise can be seen in the frequency range 5-10 kHz.

Table 10. Airflow and static pressure drop for the ATD

ATD setting	20 l/s	28 l/s
h7	82 Pa	
h10	46 Pa	82 Pa
h14	23 Pa	

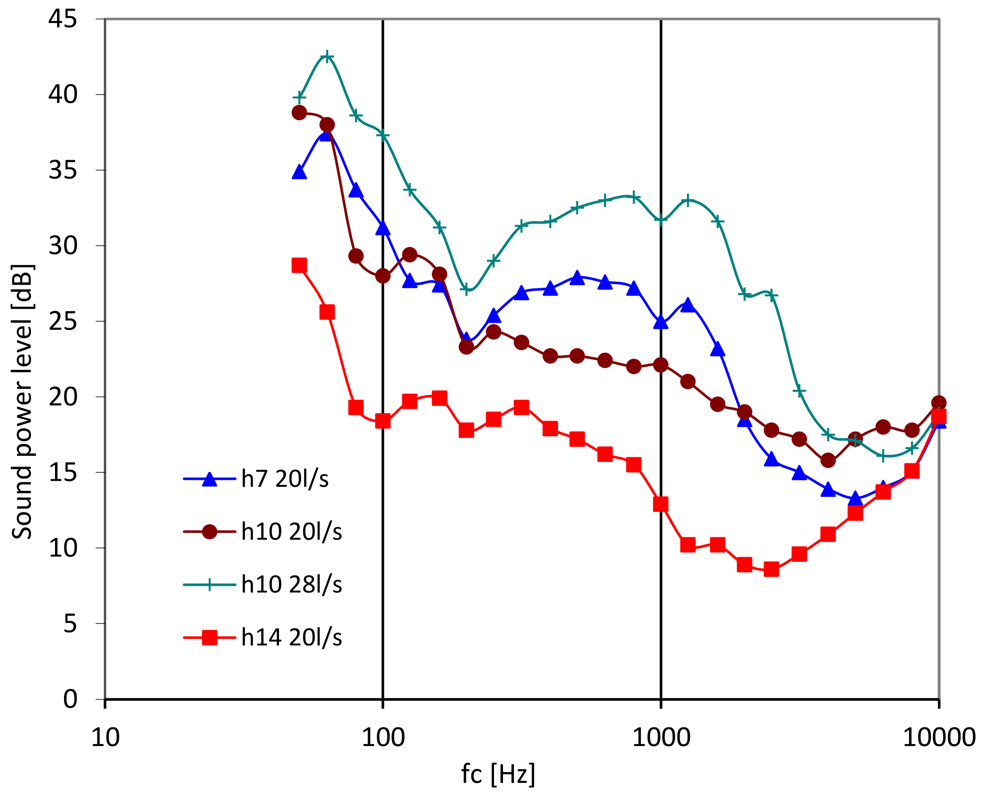

Figure 8 ATD third octave band sound power levels

To verify the noise contribution of the damper, into the receiving room, a calculation using product data is conducted. For the full system calculation see Appendix A. First, the attenuation of the silencer and ATD is subtracted from the damper flow noise. Secondly, the resulting level is compared to the HRU and ATD noise contributions in the room as presented in table 11 and 12. When the ATD setting is h10 the corresponding damper static pressure drop is 44 Pa. For this duty point the resulting noise level of the damper is always 10 dB lower than the noise level of the ATD and HRU. For the ATD setting h14 the corresponding damper static pressure drop is 67 Pa and the noise level at 63, 250 and 500Hz is less than 10 dB lower than the noise level of the HRU and the ATD. Still, the noise level in the receiving room are increased by less than 1 dB by the damper at these frequencies.

Table 11. Damper noise contribution to room (pressure drop 44 Pa and silencer 0.6m)

	63	125	250	500	1000	2000	4000	8000
Lw IRIS	50	46	41	34	29	19	13	0
Lw IRIS to room	19	24	21	16	4	-22	-19	-26
Lw HRU to room	31	38	30	27	13	-4	6	-1
Lw ATD to room	27	27	24	28	28	26	17	14
Difference	>10	>10	+10	>10	>10	>10	>10	>10

Table 12. Damper noise contribution to room (pressure drop 67 Pa and silencer 0.6m)

	63	125	250	500	1000	2000	4000	8000
Lw IRIS	54	50	44	38	32	22	16	4
Lw IRIS to room	23	28	24	20	7	-6	-14	-20
Lw HRU to room	31	38	30	27	13	-4	6	-1
Lw ATD to room	21	21	18	22	22	20	17	8
Diff.	+8	+10	+6	+8	>10	>10	>10	>10

The airflow through the main duct after the T-junction is verified by a pressure drop measurement over the nozzles of the main duct damper. The airflow, at the damper setting used, was verified using the measurement method described in section 3.4. The setting of the damper was always 6 and the corresponding pressure drop for each airflow is presented in table 13. For the HRU maximum flow, i.e. approximately 110 l/s, the pressure drop over the damper exceeds 100 Pa and the correct duty point is not possible to achieve without decreasing the pressure drop over the damper. No detailed airflow measurement was therefore conducted for the HRU maximum airflow case.

Table 13. Airflow and static pressure drop of main duct damper at setting 6

Airflow [l/s]	Component pressure drop [Pa]
50 (ref. case)	54
70	97

4.2 HVAC system noise spectra

To evaluate the possible variations of the HVAC system a number of sound measurements were conducted for the HVAC system connected to the reverberation chamber. To adjust the sound absorption of the reverberation chamber to the reference case of a living room extra absorption was introduced into the chamber. The resulting sound absorption in third octave bands and the background sound pressure level are presented in appendix C. The voltage of the RDAS supply air fan was set to 6.3 V corresponding to an airflow of 70 l/s at 100 Pa available static pressure. The airflow was validated by pressure drop measurement over the ATD at the end of the receiving room duct and the damper after the T-junction in the main duct. The static pressure was measured in the main duct before the T-junction.

The standard case, used if nothing else is mentioned, is defined using a HRU airflow of 70 l/s and an available static pressure of 100 Pa, ATD with setting h10 and an airflow of 20 l/s, no silencer at the HRU and a 600 mm long silencer in the duct to the receiving room. The mean 1/24-octave band sound pressure level in the receiving room for this standard case is presented in figure 9. The corresponding spectrum in third octave bands is, according to the simplified method in ISO 1996-2:2007 Annex D, not considered tonal. For an equivalent A-weighted level of 30 dB(A) the maximum A-weighted level does not exceed 35 dB(A). Concluding the noise as non-tonal and the difference between the maximum and the equivalent noise level less than 5 dB leaves only one legislated measure to restrict the noise level in our reference living room, i.e. an equivalent A-weighted level less than 30 dB(A).

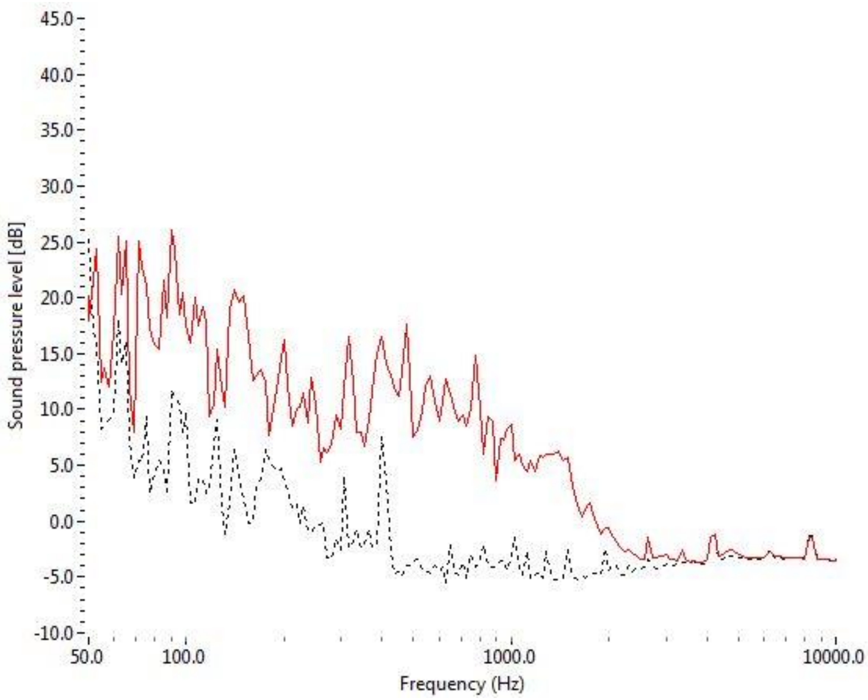

Figure 9. Sound pressure level in receiving room for standard case (red line) i.e. HRU airflow 70 l/s and ATD h10 airflow 20 l/s using a 600 mm long silencer. Background level (dashed black line)

The main variation possibility of the HVAC system, from a spectrum shape perspective, is the balance between the HRU originated noise and the ATD originated noise, i.e. approximately equivalent to low frequency noise compared to high frequency noise. The idea is to keep the total A-weighted noise level constant for all test cases and introduce variations to the spectrum shape. For these measurements most of the test cases had a similar total A-weighted noise level. When playing the sound recordings in the listening test the A-weighed level will be calibrated to the exactly same constant level for all test cases.

Figure 10 presents the spectra for three different test cases where, except from the standard case, the ATD pressure drop is decreased using setting h14 or increased using ATD setting h7. When increasing the ATD noise, i.e. setting h7, the combination of 900 and 600 mm long HRU silencers were used to further decrease the HRU originated noise. High frequency differences can be seen due to the ATD settings at 800-400 Hz and silencer attenuation at low frequencies at 80-400 Hz. The pressure drop of the combination of damper and ATD was always constant resulting in a constant static pressure in the main duct before the T-junction.

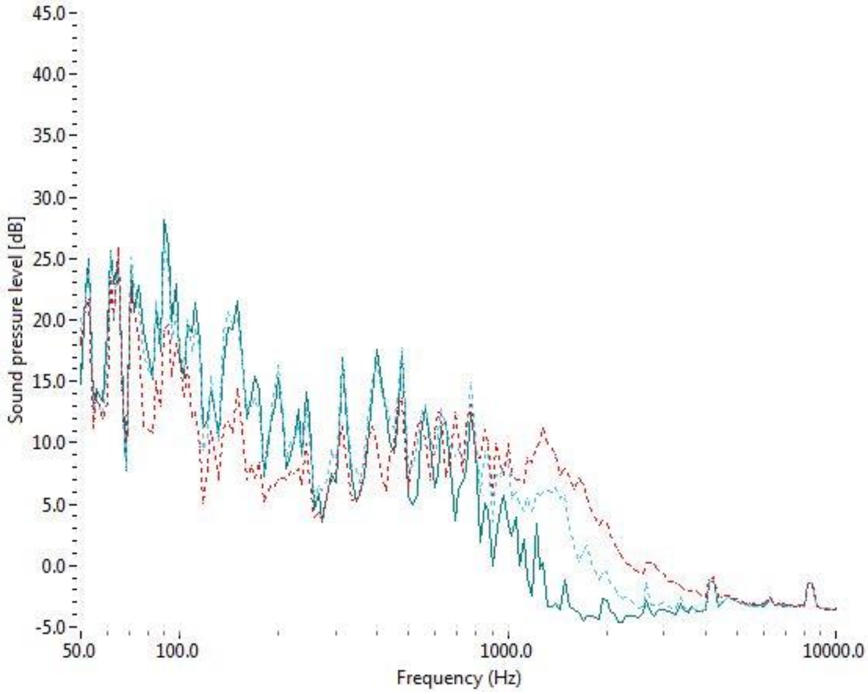

Figure 10. ATD with different settings; h14 (green), h10 (dashed light blue), h7 in combination with HRU silencers (dashed red)

Another way to cause variations to the spectrum is to introduce a flow disturbance before the ATD e.g. by a 90° bend as presented in figure 11. The test case corresponds to a common situation in modern buildings. The main difference is seen in the frequency range 500-3000 Hz. Another similar case, not tested, is a T-junction next to the ATD.

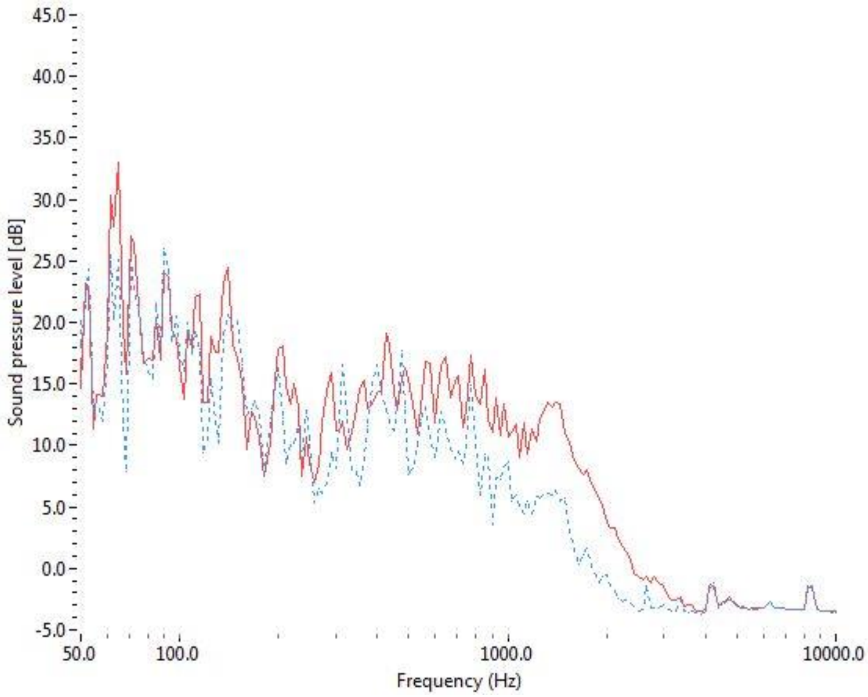

Figure 11. Bend before ATD at an airflow of 20 l/s: h10 (dashed light blue) and bend next to ATD h10 (red)

A test case that corresponds to a multi-dwelling building is to set the HRU at maximum airflow (approximately 110 l/s, denoted high airflow) but keeping the airflow through the ATD constant. Another test case, corresponding to a larger receiving room or a poorly adjusted HVAC system, is minimizing the pressure drop of the damper for the standard test case i.e. increasing the ATD airflow to 28 l/s denoted as ATD high airflow. These two test cases together with the different ATD setting test cases are presented in third octave band levels in figure 12.

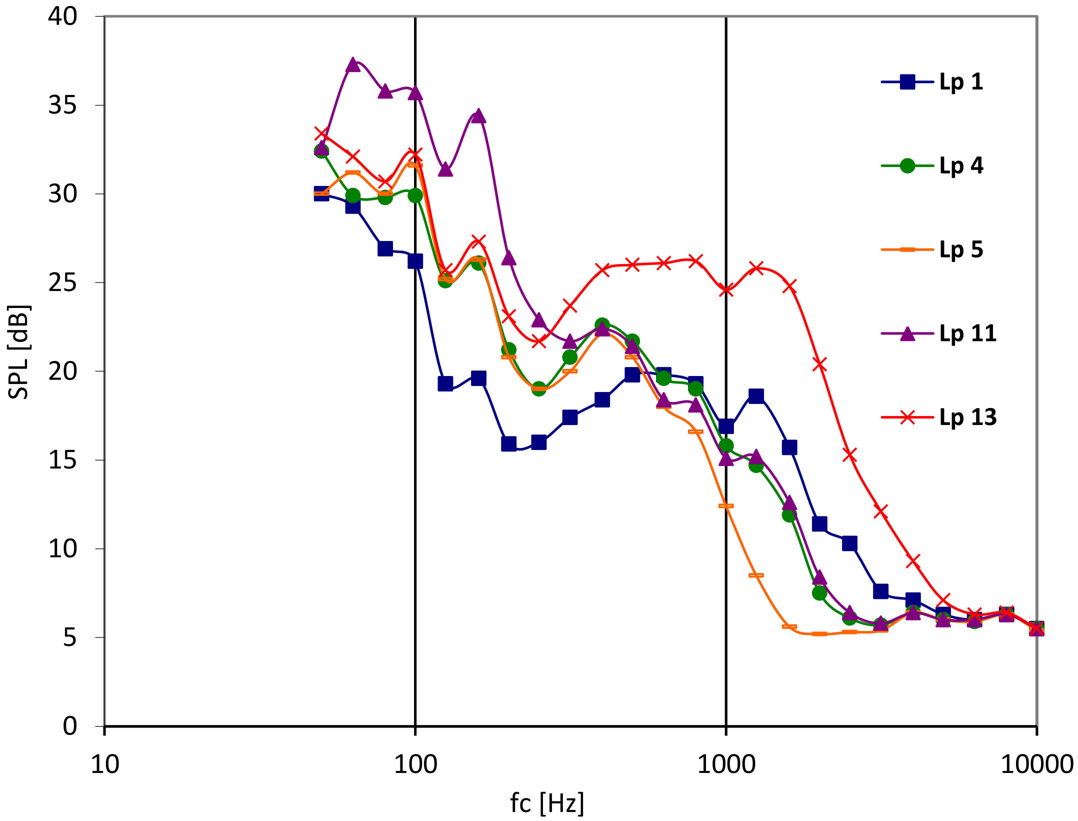

Figure 12. Third octave band sound pressure levels for HVAC system setup where ATD settings are: h7 (Lp1), h10 (Lp4), h14 (Lp5), h7 and HRU at high airflow (Lp11), h10 and ATD at high airflow (Lp13)

4.3 Recommended noise spectra for sound recordings

From the measurement results in section 4.2 a number of test cases appropriate for the sound recordings can be summarized. The variations of the ATD settings are combined with the three different HRU's as described in section 3.3. In table 14 a summary of the test cases is presented consisting of 18 combinations.

Variations of the spectrum at both high and low frequencies are present. The total A-weighted noise level is approximately constant for the standard airflows but somewhat higher for the high airflows. The level can be calibrated to a constant level during the listening test. As the spectra are considered non-tonal and the difference between the maximum and equivalent A-weighted noise levels are less than 5 dB the resulting legislated restriction of the noise is an equivalent A-weighted noise level below 30 dB(A). If the sound recordings will demonstrate large variations in experienced disturbance a need for additional measures in the recommendations for the acoustic planning of residential buildings can be concluded.

Table 14. Combinations of HVAC system settings

		HRU 1		HRU 2		HRU 3	
ATD setting	airflow	70l/s	max	70l/s	max	70l/s	max
min open	20 l/s	x		x		x	
mean open	20 l/s	x		x		x	
max open	20 l/s	x	x	x	x	x	x
Bend+ ATD mean open	20 l/s	x		x		x	
mean open	28 l/s	x		x		x	

5 Conclusion

The main conclusions for the selection of HVAC system design for the noise recordings are:

- The legislated noise criteria, e.g. for a living room, still permit variations in the noise spectrum and possible variations of the experienced disturbance.
- From a noise spectrum shape perspective the balance between the duct system components´ flow noise and the HRU generated noise is the main parameter. Low frequencies will normally be dominated by the HRU noise and the high frequencies by the flow noise from the duct components.
- Damper noise can contribute both at mid and high frequencies but the installation of a duct branch silencer will attenuate the damper noise and also prevent cross talk between rooms.
- The importance of HRU noise in each room will depend on the total airflow of the HRU, i.e. size of building to provide airflow to, and the performance of the HRU silencer
- The importance of duct component flow noise is concluded mainly dependent on the duct system layout, i.e. component pressure losses, but depends also on a correct selection of silencers.
- A poor HVAC duct system layout and system installation can result in more mid- and high frequency noise due to higher pressure loss of the duct components and a disturbed inflow to these.

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Appendix

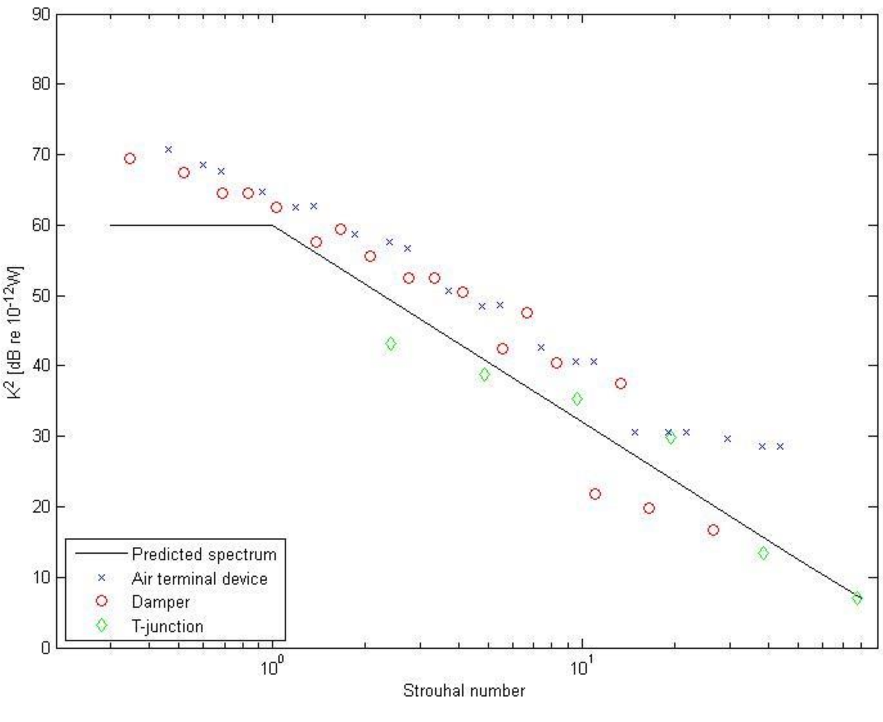
Appendix A: HVAC system noise level calculation

Lw=Sound power level, TL=Transmission Loss, FN=Flow Noise

		63	125	250	500	1000	2000	4000	8000	LwA
HRU RDAS 100Pa 70 l/s 6,3V	**Lw**	**70**	**69**	**61**	**60**	**57**	**51**	**55**	**41**	**62.9**
Silencer A 0.3m D=160, 50 mm glass wool	**TL**	**1**	**2**	**4**	**8**	**12**	**17**	**10**	**11**	
Silencer A	**FN**									
Sound power level after silencer	**Lw**	**69**	**67**	**57**	**52**	**45**	**34**	**45**	**30**	**55.6**
T-junction 160mm to100 mm	**TL**	**5.5**	**5.5**	**5.5**	**5.5**	**5.5**	**5.5**	**5.5**	**5.5**	
T-junction	**FN**									
IRIS damper	**TL**	**2**	**2**	**2**	**2**	**2**	**2**	**2**	**2**	
IRIS damper setting 6.5, 40Pa, Lp10=20dB(A)	**FN**	**50**	**46**	**41**	**34**	**29**	**19**	**13**	**0**	**37.0**
Sound power level after damper	**Lw**	**62**	**60**	**50**	**45**	**38**	**27**	**38**	**23**	**48.4**
Silencer B 0.6m D=100, 50 mm glass wool	**TL**	**12**	**9**	**13**	**18**	**25**	**37**	**27**	**21**	
Silencer B 0.6m D=100, 50 mm glass wool	**FN**									
ATD STQA-100 setting h10	**TL**	**19**	**13**	**7**	**0**	**0**	**4**	**5**	**5**	
In total from duct	**Lw**	**31**	**38**	**30**	**27**	**13**	**-4**	**6**	**-1**	**27.4**
ATD STQA-100 setting h10, 50 Pa	**FN**	**27**	**27**	**24**	**28**	**28**	**26**	**17**	**14**	**32.0**
Sound power level in total from duct and ATD	**Lw**	**32**	**38**	**31**	**31**	**28**	**26**	**17**	**14**	**33.3**
Room sound absorption		**8**	**8**	**8**	**8**	**8**	**8**	**8**	**8**	
Sound pressure level in room		**24**	**30**	**23**	**23**	**20**	**18**	**9**	**6**	**25.3**

Appendix B: Flow noise prediction of HVAC components

The flow noise generation of HVAC duct system components can be predicted using semi-empirical scaling laws. Using a reference spectrum together with component airflow, pressure drop and duct diameter results in generated sound power levels for each component. The reference spectrum of the flow generated noise using the prediction model presented in “Predicting flow-generated noise from HVAC components” [Kårekull] are here compared to recalculated product data for an air terminal device, a damper and a T-junction. Products where the octave band sound power data (125-4000 Hz) is available for specific duty points are chosen i.e. BOR-R-125 [Systemair] and IRIS-100 [FläktWoods]. T-junction data is a combination of sound power data from VDI 2081 and pressure loss data for TCPU-160-100 [Lindab]. Component duty points are chosen equivalent to the duty points of the HVAC system presented in figure 4, i.e. three duty points for ATD (20 l/s; 82, 46 and 23 Pa) and damper (20 l/s; 8, 44, 67 Pa) and one duty point for T-junction (70 l/s in and 20 l/s out in branch). The deviation between the reference spectrum and the product data exemplifies the potential accuracy of the noise prediction model. Product data is approximately within two standard deviations of the prediction model (i.e. 95% of the normally distributed results used to generate the reference spectrum).



Appendix C: Sound absorption and background noise

Third octave band sound absorption and background sound pressure level in receiving room (Fläkt Woods reverberation chamber with extra absorbers)

fc [Hz]	Sound absorption [dB]	Background Lp [dB]
50	-8,4	31,7
63	6,2	24,8
80	4,4	15,9
100	-6,0	17,3
125	7,5	13,9
160	8,8	12,5
200	8,0	12,8
250	9,0	8,5
315	8,6	8,6
400	8,1	10,4
500	8,7	5,0
630	7,9	5,0
800	8,4	5,4
1000	8,0	5,6
1250	7,7	4,6
1600	7,4	4,4
2000	7,3	4,9
2500	6,9	4,9
3150	7,4	5,2
4000	8	6,3
5000	8,7	5,7
6300	9,7	5,9
8000	10,7	6,4
10000	12,3	5,5

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